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Achieving Reliable Operation of a Steam Turbine's Automatic Control and Protection System

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Abstract—The algorithms, procedures, and modules for making expert estimates of the technical state of automatic control and protection systems for different types of turbines by means of the SPIDER mobile computerized automation system are presented. It is shown that owing to high trustworthiness with which the kind and location of hidden defects of units are determined without disassembling them, repair works can be scheduled in the optimal way and the costs for carrying them out can be reduced.

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The majority of steam turbines in Russia are equipped with hydromechanical and hydrodynamic automatic control and protection systems (ACPSs), which are susceptible to obsolescence and physical wear. High-quality and reliable operation of these turbines complying with the requirements of [1] under the conditions of shifting to the system of carrying out repairs of turbine units in accordance with their technical state can be achieved owing to the use of high-efficient and high-performance means of technical diagnostics that make it possible to reveal hidden defects in ACPS components and improve their design in the subsequent. The mobile computerized automation system (CAS) SPIDER [2] constructed on the basis of high-tech instrumentation-grade equipment produced by HBM Mess & Systemtechnick GmbH (Germany) and certified by the State Agency of the Russian Federation for Standardization (Gosstandard) is one of such means. The required depth of diagnostics is achieved due to high precision of the used instrumentation and algorithms for calculating diagnostic criteria, and high performance is achieved due to fully automated acquisition, processing, and analysis of data. The healthiness of ACPS components is determined using the SPIDER computerized automation system by pressing a single pushbutton immediately after completing a test carried out directly on the turbine. The turbine ACPS is diagnosed by comparing the following indicators with their permissible values:

(i) whether a parameter goes out from the specified range;

(ii) whether functional and time dependences go out from the region bounded by the upper and lower tolerances;

(iii) the transfer ratio $\Delta y/\Delta x$ (the slope of the characteristic);

(iv) the rate of change of a parameter dy/dt;

(v) irregularity and degree of irregularity;

(vi) the steady peak-to-peak ripple Z_{st} of a parameter; and

(vii) insensitivity or degree of insensitivity.

Separation of dynamic insensitivity into static and inertial components and development of algorithms for determining them are the most important factors due to which high effectiveness of practical applications of the SPIDER system is achieved [3].

Insensitivity manifests itself as lack of uniqueness of the values of an output quantity vs. a change in the input quantity when control members are displaced in one direction or another. Two kinds of insensitivity are distinguished: absolute insensitivity Δ and reduced degree of sensitivity ε. In diagnosing ACPS components, these characteristics are determined from the frequency (Δ_n, ε_n), displacement (Δ_h, ε_h), pressure (Δ_p, ε_p), and electric power (Δ_N, ε_N) in accordance with the test method (a two-pass static characteristic or a two-pass dynamic characteristic).

At present, manufacturers present standard data on static insensitivity [Δ_{st}], degree of static insensitivity [ε_{st}], and steady peak-to-peak ripple [Z_{st}] obtained using the traditional (manual) methods.

ACPS components are called inertialess if the output quantity begins to change at the same moment with the change in the input quantity; i.e., the delay time $t_{del} = 0$. For such components, both static and dynamic insensitivities are uniquely determined:

$\Delta = \Delta_{st}$ (the method of two-pass static characteristics), and

$\Delta = \Delta_{dyn} = \Delta_{dyn(t_{del}=0)}$ (the method of two-pass dynamic characteristics).

Theoretically, the following holds for inertialess components: $\Delta_{st} = \Delta_{dyn} = \Delta_{dyn(t_{del}=0)}$.

In low-inertia and inertial ACPS components (the former have a closing time of no more than 0.08 s and the latter have a closing time of no less than 0.2 s), the moment at which the output quantity begins to change does not coincide with the moment at which the input quantity begins to change; i.e., $t_{del} \neq 0$. For such components, their static insensitivity is uniquely determined, and Δ_{inert} is one of the components of the total dynamic insensitivity:

$\Delta = \Delta_{st}$ (the method of two-pass static characteristics); and

$\Delta = \Delta_{dyn} = [\Delta_{dyn(t_{del}=0)} + \Delta_{inert}]$ (the method of two-pass dynamic characteristics).

The insensitivity data obtained from the static and dynamic tests are used to determine the inertial component of the dynamic insensitivity:

$$\Delta_{inert} = \Delta_{dyn} - \Delta_{st}.$$

If the standard Δ_{st} is taken as the standard Δ_{dyn} for low-inertia components, this will be a factor making the requirements for their technical characteristics more stringent.

In practical applications, it is rather difficult to determine $\Delta_{n\,st}$ with which the turbine frequency is controlled, and only the method of two-pass dynamic characteristics is applicable for it:

$$\Delta_n = \Delta_{n\,dyn} = [\Delta_{n\,dyn(t_{del}=0)} + \Delta_{n\,inert}].$$

Thus, the dynamic insensitivity $\Delta_{dyn(t_{del}=0)}$ can be uniquely determined only for inertialess components, and for inertial components this insensitivity is one of the components of the total dynamic insensitivity Δ_{dyn} and is determined during the static test. It is important to note that specialists in the subject area are interested in exactly this insensitivity as a parameter most descriptively telling about the presence and value of friction forces in the system. At the same time, inertia is an objective reality, and the component $\Delta_{n\,inert}$ in the total insensitivity $\Delta_{n\,dyn}$ is important, e.g., for analyzing how the ACPS responds to a deviation in the network frequency, which should more correctly be represented as a "sensitivity threshold."

Below, we consider the algorithms, test procedures, and results obtained from diagnosing the main components of different types of ACPSs for steam turbines carried out using the SPIDER computerized automation system.

TIMING PARAMETERS OF AN ACPS

According to the requirements specified in sections 4.4.10 and 4.4.11 of the Operational Regulations [4], it is necessary to check the operation times of individual components and the ACPS as a whole. The design values of the total closing time T_{tot}, self closing time T_{self}, delay time of components T_{del}, and rotor inertia constant T_{rot} are given in the turbine certificate. Tests for automatically determining slide valve closing times, servomotor traveling times, and rotor inertia constant using a specially developed algorithm are integral parts of ACPS diagnosing procedures [5]. The test is carried out using the method of single-pass dynamic characteristics. In doing so, the start and stop flip-flops, and the main and auxiliary channels are specified. The main channels correspond to the assemblies the characteristic times of which are determined. The auxiliary channel is required for setting up a zero time reference point. For slide valves and servomotors, this point usually corresponds to the moment at which the overspeed governor's slide valve (OGSV) starts to move, or to the moment at which the frequency begins to increase in carrying out the instantaneous load rejection test for the turbine rotor. To record the beginning of motion, the accuracies A and B of sensors in the auxiliary (X) and main (Y) channels are specified (for the OGSV and other slide valves, $A = 0.05$ mm; for servomotors, $B = 0.2$–0.3 mm; and for the rotor, $B = 0.01$ Hz). If the numerical values of the parameters of the auxiliary and main channels decrease during the test, A and B are specified with the "plus" sign, and if they increase, A and B are specified with the "minus" sign.

With the turbine held in the shutdown state, the ACPS components are set in their design (initial) positions by manipulating with the turbine speed changer (TSC) and the heat load changer (HLC), after which these components are closed by pressing the pushbutton for manually tripping the turbine (MTTP). The SPIDER computerized system calculates the closing times and compares the obtained values with the certificate data. There is an additional possibility to calculate the slope of the experimental dependence. The calculation algorithm makes it possible to determine the slope of the curve, the motion start moment or time delay, the motion time, and the moment at which the motion is finished. The labels (numbers) of points at which the initial values are recorded are determined by the quantities A and B. For the auxiliary channel, the point $t = 0$ is set based on the deviation from the initial value by the sensor accuracy A (Fig. 1); and for the main channel, the length of the horizontal section t_1 (from the point $t = 0$) is determined based on the deviation from the initial value by the sensor accuracy B. The final value of displacement for the main channel is calculated as the mean value of the last recorded measurements. The time corresponding to the end of the inclined section t_2 (stop) from the moment $t = 0$ is

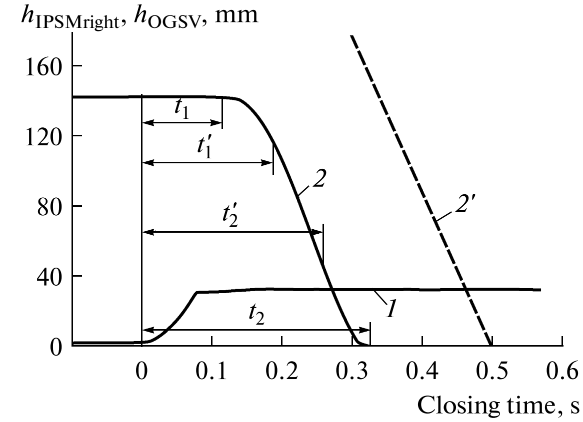

Fig. 1. Curves characterizing the closing time of the right-hand intermediate-pressure servomotor (IPSMright) used in the T-250/300-23.5 turbine of TMZ. (*1*) Overspeed governor's slide valve (OGSV), (*2*) IPSMright, and (*2*') reference curve for IPSMright.

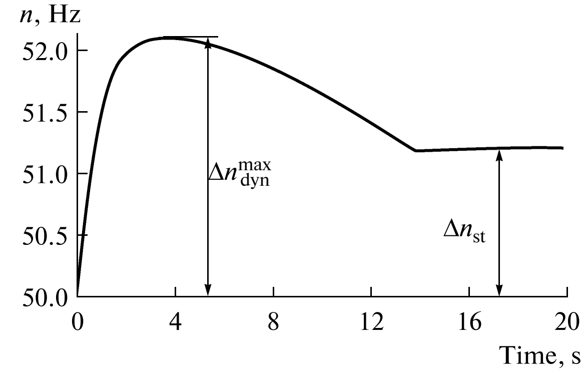

Fig. 2. Curve characterizing instantaneous rejection of the electrical load $N = 2.05$ MW of the P-6-1.6/05 turbine produced by the Kaluga Turbine Works.

determined. To do so, the sensor accuracy B is added to the displacement final value, and the moment is found at which a lower value is recorded for the first time for the main channel. After that, the following parameters are calculated depending on the algorithm:

$$\text{slope } [y(t_1') - y(t_2')]/(t_2' - t_1');$$

$$\text{motion start time } t_1 = T_{\text{del}};$$

$$\text{motion time } t_2 - t_1 = T_{\text{self}}; \text{ and}$$

$$\text{motion end time } t_2 = T_{\text{tot}}.$$

The initial and final values are calculated as mean values from the first and last buffers of measured quantities. In calculating the slope, indents are made from the beginning and end of the inclined section by approximately one-third of it: t_1' and t_2' (see Fig. 1).

After an instantaneous rejection of electrical load, the maximal dynamic $n_{\text{dyn}}^{\max}$ and static Δn_{st} rotor frequency increase values are determined, the rotor inertia constant is calculated, and the control process quality is estimated (Fig. 2).

OVERSPEED GOVERNORS AND RS-3000 CENTRIFUGAL FREQUENCY CONTROLLERS

The mechanical overspeed governor (OG) is at present the main and the most reliable element of the turbine rotor overspeed protection. According to the requirements specified in section 4.4.5 of the Operational Regulations [4], and for putting in use the procedure for diagnosing a mechanical OG without speeding up the turbine that was proposed by specialists of the All-Russia Thermal Engineering Institute (VTI) [6], a universal automated setup with the possibility of oil-driven testing of the OG strikers has been constructed, which maximally simulates the conditions of their operation. The setup makes it possible to carry out calculated and experimental research works on studying the influence of vibration and friction forces on the stability of striker actuation, and the dependence of spring steel elasticity modulus on the temperature and acceleration (rate) with which the turbine rotor is speeded up.

The 11-kW AIRM 132 M2 three-phase asynchronous motor for running the setup is manipulated by means of the controller used in the L300P-110HF variable-frequency drive produced by Hitachi. The SPIDER computerized system, using which the striker actuation and setback frequencies are recorded, serves as a measurement facility. This system is used in tuning a mechanical OG on the setup and in diagnosing it on a turbine, due to which high trustworthiness of an analysis of its technical state is achieved.

The speed and accuracy with which the striker actuation frequency is measured are extremely important for estimating the technical state of the strikers. The 1GT-101-DC frequency sensor is installed on a gear wheel with 60 teeth with a gap between them equal to 2–4 mm and produces the signals at its output as a protrusion or an indent passes by. The maximal theoretically possible frequency of the turbine rotor is equal to 100 Hz. Sixty teeth pass through the sensor per revolution; therefore, the sensor will record 120 changes of voltage level. If only the leading edges of pulses are taken into account, 60 pulses are generated per revolution and, accordingly, 6000 pulses per second. The SPIDER's measuring amplifier converts such a pulse sequence with accuracy class 0.01. The end-to-end channel with a digital output, which records pulses in accordance with a certain condition, has a time delay equal to 1 ms.

The secondary transducer records input pulses having a pulse ratio equal to 2 with a time delay of 2 μs (the pulse ratio here is understood to mean the ratio of

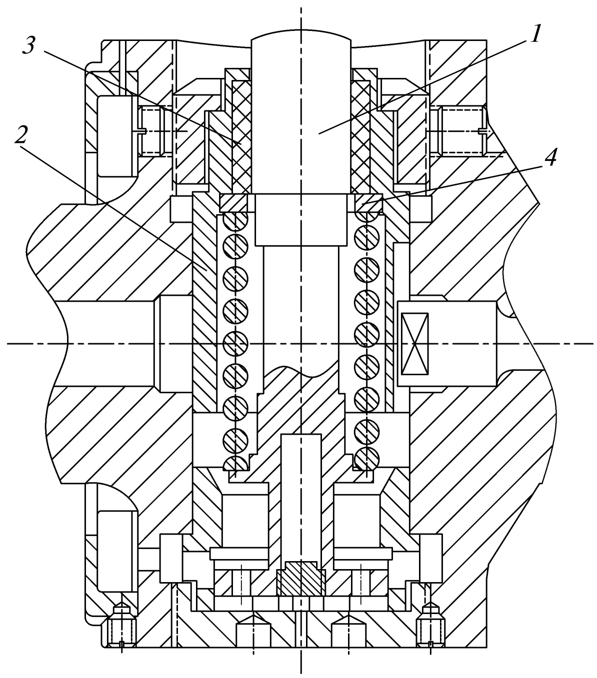

Fig. 3. Modernization of the overspeed governor used in the PT-80/100-12.8 turbine of LMZ. (*1*) Finger-type striker, (*2*) bronze bushing, (*3*) fluoroplastic insert, and (*4*) ring.

the total width of the tooth and indent to the tooth width). The sensor actuation delay times are equal to 15 μs for the pulse leading edge and 1 μs for the pulse trailing edge. The minimal duration of the pulse at the maximal frequency equal to 60 Hz per tooth of the gear wheel is equal to 160 μs. Thus, the SPIDER computerized system allows the pulses generated in the turbine rotor frequency measurement system to be recorded in a guaranteed manner.

The actuation and setback of strikers are recorded using two other 1GT-101-DC frequency sensors. The installation gap of these sensors is selected to be larger than their sensitivity range and the maximal stroke of the striker so that the sensor's output signal was generated only when the striker has come into action [7].

The overspeed governor is dismantled from the turbine, installed in the setup, and balanced so that the vibration of supports does not exceed 30–40 μm at a frequency of 55 Hz. The medium (finely dispersed dust) in which the OG operates on the turbine is simulated, and two tests with different speedup rates are carried out. The speedup rate in the first test corresponds to the value indicated in the manufacturer's operating manual, and the speedup rate used in the second test corresponds to the maximal acceleration, which is determined by the rotor inertia constant defined as the period of time for which the frequency increases by 50 Hz at the rated parameters of steam upstream and downstream of the turbine supplied to it with the nominal flowrate. The dependence of the striker actuation frequency on the rotor speedup rate has been established experimentally. The way in which the inertia forces influence the striker's angular actuation frequency follows from the formula

$$\omega_{\mathrm{act}} = [(ch_{\mathrm{pr}} - F_{\mathrm{in}})(m_{\mathrm{str}}e)]^{1/2},$$

where c is the spring stiffness ratio, h_{pr} is the spring preliminary compression, F_{in} is the inertia force, m_{str} is the striker mass, and e is the eccentricity or the striker center of gravity gyration radius.

The frequency at which the striker comes into action in speeding up the turbine tends to decrease as the rotor acceleration increases. This means that the mechanical overspeed governor has a self-regulation feature. Owing to this feature of the mechanical OG and the function of analyzing the dynamics of the parallel electronic OG, essentially better reliability of the protection is achieved when full electrical load is instantaneously rejected to zero.

After the tests, the OG is disassembled, subjected to instrument-assisted diagnostics, and its worn parts are replaced. The strikers are adjusted again after the repair, the characteristics of the striker actuation and setback frequencies are taken during speedup tests and the striker actuation frequencies during oil filling tests vs. the rod or adjustment nut rotation angle are determined. The status of reference dependences is assigned to these characteristics, and they are entered in the TURB folder of the SPIDER system software for use in diagnosing the OG on the turbine. An electronic repair form is filled in, which contains fields for entering the results from instrument-assisted diagnostics and tests of strikers by speeding up the turbine with two rates and by filling oil before and after the repair. Hard copies of the above-mentioned documents are forwarded to the customer together with the OG unit. The results obtained from the tests of the mechanical OG at the setup can be used for programming the setpoints of the regular electronic overspeed governor used in the turbine ACPS. With the turbine running in the idle mode, the striker actuation frequencies are determined from the oil filling tests. If these frequencies coincide with the results of tests on the setup, the rotor speedup test is not carried out. The results obtained from subsequent striker tests carried out by oil filling are compared with the preceding results, and the technical state of the strikers is determined. If a lower actuation frequency is obtained, this means that the spring has settled or that the spring steel's elasticity modulus has decreased; if a higher actuation frequency is obtained, this means that additional friction forces have appeared. If jams occur, gradual (static) wearing-out of the striker may be observed [9].

Poor performance of the turbine rotor current pickup brushes causes damage to a finger-type striker (Fig. 3) and bronze guide bushing used in OGs

installed on the turbines produced by the Leningrad Metal Works (LMZ). This leads to a failure of the overspeed protection, and a few emergency shutdowns (followed by testing the strikers in the speedup mode) have to be made for removing this failure during the period between planned turbine repairs. The OGs installed on a few PT-80/100-12.8 turbines have been modernized on agreement with LMZ. The idea of this modernization is that a fluoroplastic insert is installed in the guide bushing bore and is fixed there by a ring calked over the circumference. The use of this technical solution ensures trouble-free and stable operation of the OG for more than 5 years.

The RC-3000 rotation speed regulator is attached to the setup shaft instead of the overspeed governor, and a WA-20 displacement sensor is installed on the turnback plate. Fully automated tests are carried out using the methods of two-pass static and dynamic characteristics. The obtained characteristic of the RS-3000 regulator and criteria are entered in the TURB folder in the SPIDER system software for the given turbine as standard ones and are used in the subsequent for diagnosing the ACPS.

SLIDE VALVES AND SERVOMOTORS

That the insensitivity and ripple of a slide valve and servomotor are in line with their standard values serves as an indicator of their high-quality and reliable operation [10]. By slide valves we mean intermediate slide valves, slide valves used in the frequency and pressure controllers, slide valves of the electrohydraulic converter, and cutoff valves of servomotors. Insensitivity is caused by resistance forces, and ripple is caused by resistance forces and by the forces of sign-variable impulse disturbances. If these disturbances have a quasi-steady pattern, an increased insensitivity resulted from ingress of nontrapped particles into the gap between the journal boxes and pistons of slide valves and servomotors causes this ripple to become smaller. A numerical value of only one of the criteria does not give one grounds to consider that the slide valve and servomotor have a healthy state even if its value does not exceed the standardized level. Poor metrological performance of traditional instrumentation used for carrying out tests and adjustment under the conditions of a power station do not make it possible to determine the insensitivities of rotating slide valves used in oil- and water-operated ACPSs produced by the Turbine Engine Works (TMZ) from the impeller pressure and compare these insensitivities with their standard values. Thus, for the slide valves used in the speed governor of T-110/116-12.8 and T-250/300-23.5 turbines, $[\Delta_{p\,\mathrm{st}}] \leq 0.003$ MPa, and for the intermediate slide valve (ISV), $[\Delta_{p\,\mathrm{st}}] \leq 0.004$ MPa at $[Z_{h\,\mathrm{st}}] \leq 0.4$ mm. For specialists who adjust ACPSs it is preferable to have data on the insensitivity of these slide valves for displacement, because the impeller

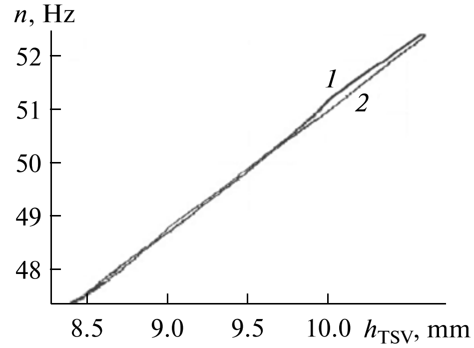

Fig. 4. Curves characterizing the tracking system of the PT-80/100-12.8 turbine of LMZ (the method of two-pass dynamic characteristics). (*1*) Direct stroke and (*2*) back stroke.

pressure is the input signal, and displacement is the output signal. Correlations between the insensitivity and ripple of rotating speed governor slide valves (SGSVs) and intermediate slide valves (ISVs) used in the ACPSs of TMZ turbines have been experimentally determined and can be used as reference ones, because firmly high-quality and reliable operation is achieved if these correlations are kept. For a T-110/116-12.8 turbine, $\Delta_{h\,\mathrm{SGSV}} \leq 0.1$ mm and $Z_{h\,\mathrm{st\,SGSV}} \leq 0.2$ mm, and for a T-250/300-23.5 turbine, $\Delta_{h\,\mathrm{SGSV}} \leq 0.06$ mm, $Z_{h\,\mathrm{st\,SGSV}} \leq$ 0.1 mm, $\Delta_{h\,\mathrm{ISV}} \leq 0.08$ mm, and $Z_{h\,\mathrm{st\,ISV}} \leq 0.15$ mm.

The low-inertia tracking slide valve (TSV) used in the frequency control loop of the PT-80/100-12.8 turbine produced by LMZ tracks the position of the RS-3000 speed governor rigidly connected to the shaft of the main oil pump (MOP), which is connected with the high-pressure rotor (HPR) through a tooth-type coupling. In taking the speed characteristic using the method of a two-pass dynamic response, we determine the insensitivity of the MOP–HPR–RS-300–TSV tracking system rather than the tracking slide valve itself, although the displacement sensor is installed on this valve:

$$\Delta_{h\,\mathrm{t.s}} = \Delta_{h\,\mathrm{MOP\text{-}HPR}} + \Delta_{h\,\mathrm{RS\text{-}3000}} + \Delta_{h\,\mathrm{TSV}},$$

where $\Delta_{h\,\mathrm{t.s}}$ is the insensitivity of the tracking system, $\Delta_{h\,\mathrm{MOP\text{-}HPR}}$ is the insensitivity of the coupling between the MOP and HPR, which can be equal to the axial freedom of the MOP shaft (0.05–0.07 mm), $\Delta_{h\,\mathrm{RS\text{-}3000}}$ is the insensitivity of the RS-3000 speed governor equal to 0.01–002 mm, and $\Delta_{h\,\mathrm{TSV}}$ is the insensitivity of the tracking guide valve.

Deviation of the oil pressure difference across the TSV piston and an increase of $\Delta_{h\,\mathrm{t.s}}$ above the standardized value of $\Delta_{h\,\mathrm{SG}}$ in the absence of jamming the MOP coupling can serve as indicators of jamming the tracking slide valve. Jamming of the MOP coupling can be diagnosed from the fact that $\Delta_{h\,\mathrm{t.s}}$ is equal to the sum of the MOP shaft axial freedom and $\Delta_{h\,\mathrm{RS\text{-}3000}}$ when there is no jamming in the tracking slide valve.

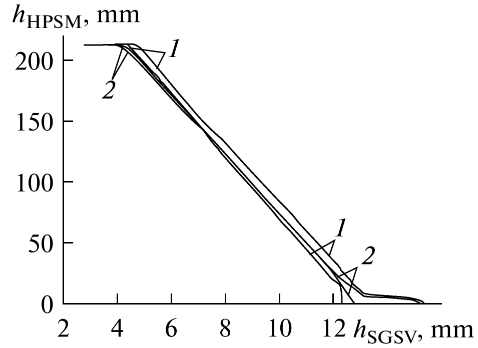

Fig. 5. Stroke of the high-pressure servomotor (HPSM) vs. the stroke of the speed governor's slide valve (SGSV) 70-mm in diameter (from the lower stop) of the T-110/116-12.8 turbine of TMZ. (*1*) Method of two-pass dynamic characteristics and (*2*) method of two-pass static characteristics.

Below, the tracking system characteristic is considered in the range n = 47.5–52.5 Hz (Fig. 4). As is known, the axial freedom of the MOP shaft measured before the startup is equal to 0.06 mm, the beating of the RS-3000 speed governor's turnback plate at an 8-mm diameter is equal to 0.01 mm, and the RS-3000 insensitivity determined at a test rig is equal to 0.02 mm.

An analysis of the results shows that the dynamic insensitivity the tracking system has at n = 50 Hz is equal to 0.023 mm, and its insensitivity for rotation frequency at a stroke of 9.556 mm is equal to 0.042 Hz (the standardized value is equal to 0.05 Hz), which points to the fact that the MOP has good alignment with the HPR, and that there are no jams in its coupling. Local jamming of the tracking slide valve and the MOP coupling is observed during a direct stroke in the frequency range 50.67–52.33 Hz ($\Delta_{h\,\mathrm{t.s}}$ = 0.081 mm, which is larger than the sum of the MOP shaft's axial freedom and the insensitivity of RS-3000), which disappeared in the back stroke.

Since servomotors are inertial elements, the method of two-pass static characteristics is used for determining their insensitivity. If we determine the insensitivity for the high-pressure servomotor used in the T-110/116-12.8 turbine of TMZ ($\Delta_{h\,\mathrm{HPSMdyn}}$) by applying a continuous control input to the single-turn electrical mechanism of the turbine speed changer toward opening and closing, it turns out that this insensitivity is a few times larger than $\Delta_{h\,\mathrm{HPSMst}}$ (Fig. 5). Thus, if the speed governor's slide valve with a diameter of 70 mm has a stroke equal to 5.8 mm, $\Delta_{h\,\mathrm{HPSMst}}$ = 1.2 mm (at $[\Delta_{h\,\mathrm{HPSMst}}] \le 2$ mm) and $\Delta_{h\,\mathrm{HPSMdyn}}$ = 7.3 mm. The inertial component of the dynamic insensitivity determined from the formula

$$\Delta_{h\,\mathrm{HPSMinert}} = \Delta_{h\,\mathrm{HPSMdyn}} - \Delta h_{\mathrm{HPSMst}}$$

is equal to 6.1 mm.

STEAM ADMISSION SYSTEMS OF STEAM TURBINES EQUIPPED WITH CAM-TYPE DISTRIBUTION DEVICES

The high- and intermediate-pressure steam admission systems (SASs) of steam turbines consist of one or two units (left- and right-hand ones) [11], each comprising a servomotor of control valves (CVSM), a cam-type distribution device (CDD), and a few control valves (CVs). As the valves are opened, the compression of their springs and the pressure difference across the CVSM piston (which is equal to the difference of the pressures under and above the CVSM piston) increase and decrease as the valves are closed. The resistance forces existing in the CVSM, CDD, and CVs are a factor that makes the dependence of pressure difference on the CVSM stroke Δp (CVSM) steeper, and settling of the CV springs makes this dependence flatter. Thus, a deviation of Δp (CVSM) from its reference characteristic is a universal indicator of the unhealthy state of the entire SAS. Manufacturers do not give reference dependences for Δp (CVSM), which adds difficulty to diagnostics of SASs. In this case, the dependence of Δp (CVSM) taken after a SAS overhaul can be taken as the reference one. The insensitivity of an inertialess SAS in displacement can be represented as follows:

$$\Delta_{h\,\mathrm{SAS}} = \Delta_{h\,\mathrm{CVSM\,disc}} + k_1\Delta_{h\,\mathrm{CDD}} + k_2\Delta_{h\,\mathrm{CV}},$$

where k_1 and k_2 are proportionality coefficients, $\Delta_{h\,\mathrm{CVSM\,disc}}$ is the insensitivity of the CVSM disconnected from the CDD, $\Delta_{h\,\mathrm{CDD}}$ is the insensitivity of the CDD, and $\Delta_{h\,\mathrm{CV}}$ is the sensitivity of the CVs.

Jams existing in a SAS show themselves on the running turbine when a stepped change occurs in the load, or when the CVSM stops in an intermediate position at a pressure under its piston close to the maximal value. The traditional approach used to determine the unit in which jamming is observed (when an increased pressure difference across the servomotor's piston is revealed) consists in that the valves are taken out from operation one by one by increasing the gap between the roller and cam or by removing the lever, or by disconnecting the CVSM from the CDD, which involves additional repair costs. One of the aims of diagnosing a SAS is to locate resistance forces in an element-wise manner. The main static and dynamic test methods and an additional dynamic test method are used. In the main method, the valves are opened (direct stroke) and closed (back stroke) by applying a control command on the speed changer's single-turn electrical mechanism, and the reference and experimental dependences of the CV stem strokes and Δp on the CVSM stroke are visualized. It is more preferable to use the dynamic test, because it shows the process pattern in a more vivid form. If jams occur in the SAS along the CVSM stroke, steps appear on the dependence of Δp (CVSM), which can be correlated, as the

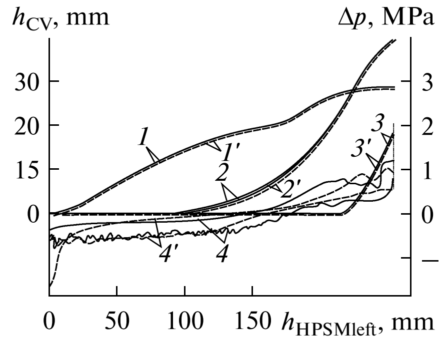

Fig. 6. Steam admission characteristics of the left-hand unit of the T-250/300-23.5 turbine of TMZ (the method of two-pass dynamic characteristics). (*1*) and (*1*') Data on CV No. 2 obtained from the preceding and next tests, (*2*) and (*2*') the same for CV No. 4, (*3*) and (*3*') the same for CV No. 6, and (*4*) and (*4*') the same for Δp.

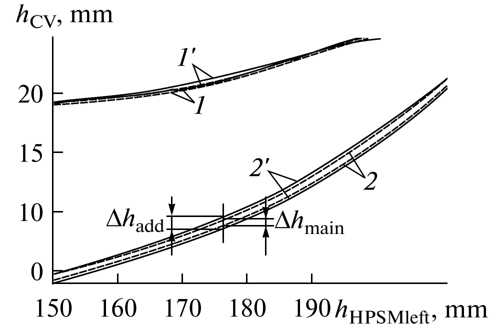

Fig. 7. Steam admission characteristics of the left-hand unit of the T-250/300-23.5 turbine of TMZ (the method of two-pass dynamic characteristics). (*1*) and (*1*') Data on CV No. 2 obtained using the main and additional methods, and (*2*) and (*2*') the same for CV No. 4.

next valve starts to open (close), with local or global insensitivity, or with insensitivity of the CDD, which also can be local or global. Local jamming is characterized by the fact that the stepped deviation of the experimental Δp from its reference value appears and disappears, and if there is global jamming, this step persists during the entire stroke of the CVSM. In frequent occasions, the SAS is not repaired due to insignificant jams and due to a limited period of time for which a turbine is taken for outage, and in the subsequent, the development of the defect is traced by comparing the above-mentioned dependences obtained in the preceding and next tests drawn on the same graph (Fig. 6).

According to the additional method for carrying out a dynamic test, the CVs are opened from the speed changer and closed from the CDD. In this test, it is possible to obtain the most salient manifestation of valve insensitivity (Fig. 7) and simultaneously determine the closing time of both the CVs and CVSM.

As was mentioned above, if there are no jams in the SAS, a reduced pressure difference across the CVSM piston may be due to settling of the CV springs. An increased CV closing time is a sign pointing to settling of its springs. The CVSM closing time does not exceed its standard value if there are no jams in it and in the CDD. Jams in the CV usually do not cause an increase in the CVSM closing time. The CV and CVSM closing times are determined using the algorithm described above. In diagnosing the CV on a shutdown turbine, only one element of the frame–stem–valve chain is checked, because the design features are such that a linear displacement sensor can be installed only on the frame. For determining the extent to which the valve stem suspension elements are worn on the running turbine, it is possible to use an algorithm based, for example, on analyzing the change with time of steam flow pulsations downstream of the CV and in the control stage chamber [12]. The following signs are laid down in the module for estimating the technical state of a SAS by expert methods:

(i) If the pressure difference across the CVSM piston corresponds to its standard value, the SAS is in a healthy state provided that the CV does not have insensitivity.

(ii) If the pressure difference across the CVSM piston does not correspond to its standard value and the CV does not have insensitivity, there is some defect in the CVSM–CDD system.

(iii) If the pressure difference across the CVSM piston does not correspond to its standard value and the CV has insensitivity, and the CVSM has closing times larger than their standard values, there are defects in the CV and in the HPSM–CCD system.

(iv) If the total closing time and insensitivity of the CVSM disconnected from the CDD do not correspond to their standard values, there is a defect in the CVSM.

(v) An increased amplitude of steam pressure pulsations downstream of the CV above its limiting value means that the stem suspension elements are worn.

(vi) If the steam pressure downstream of the CV is equal to that in the control stage chamber, this means that the valve stem is broken.

Experience gained from diagnosing the SASs used in different types of turbines shows that malfunction of the HPSM–CCD subsystem is determined in the overwhelming majority of cases only by malfunction of the CCD. The procedure for diagnosing SASs is universal in nature and can be used both for traditional hydraulic and for modernized microprocessor electrohydraulic ACPSs.

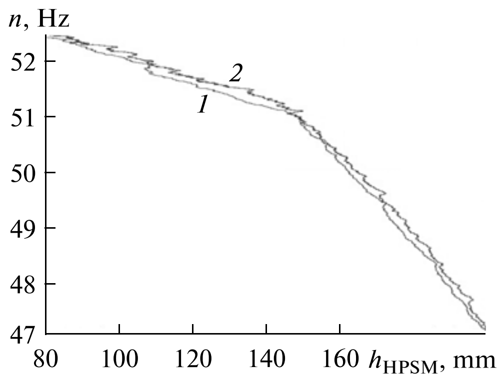

Fig. 8. Curve characterizing the control of rotation frequency of the T-110/116-12.8 turbine of TMZ (the method of two-pass dynamic characteristics). (*1*) Direct stroke and (*2*) back stroke.

DETERMINING THE INSENSITIVITY OF TURBINE FREQUENCY CONTROL

With the turbine running in the idle mode, the frequency control characteristic is taken using the methods of two-pass dynamic characteristics using the bypass of the main steam gate valve. This method is used because it is very difficult to take static characteristics due to operational limitations. A continuous control input is applied to the bypass by means of an electric drive controlled from the control board and having a fixed motion speed in the direct and back strokes. The speed changer position remains unchanged in taking this characteristic. The insensitivity in frequency n is determined as follows:

$$\Delta_{n\,\mathrm{dyn}} = \Delta n_{n\,\mathrm{dyn}(t_{\mathrm{del}} = 0)} + \Delta_{n\,\mathrm{inert}}.$$

For the T-110/116-12.8 turbine of TMZ, it has been obtained that $\Delta_{n\,\mathrm{dyn}}^{\max} = 0.23$ Hz at $[\Delta_{n\,\mathrm{st}}] \le 0.15$ Hz (although this comparison is not correct) (Fig. 8).

In reality, the insensitivity of turbine frequency control can be determined only by the method of two-pass dynamic characteristics. The slower the rate with which the turbine rotor frequency is changed during the test, the closer this characteristic to the static insensitivity.

For determining the insensitivity components of frequency control in the absence of static test data, it is necessary to develop an algorithm that would make it possible to unequivocally determine the values of $\Delta_{n\,\mathrm{dyn}(t_{\mathrm{del}} = 0)}$ and $\Delta_{n\,\mathrm{inert}}$ from the results of the performed dynamic test. One version of the algorithm for calculating the insensitivity components of these quantities using the method of two-pass dynamic characteristics can be constructed on the basis of the mathematical models of ACPS components and assembly units [13] with incorporating a high-precision generator of normalized test signal applied to the system input, the use of which will make it possible to check the ACPS also during operation under load without making a considerable change in the turbine operating mode. The means for acquiring and processing data available in the SPIDER computerized system enable the developer to revise and considerably extend the set of criteria for carrying out more accurate diagnostics of ACPSs.

CONCLUSIONS

(1) Elements of an advanced integrated system constructed on the basis of the SPIDER computerized system have been developed, which is aimed at ensuring reliable operation of ACPSs under the conditions of repairing turbine units in accordance with their technical state.

(2) The results obtained from diagnostics of the components of ACPSs installed in different types of turbines that is carried out by means of the SPIDER computerized system can be used for extending the bases of standard characteristics at manufacturing plants, and the algorithms for calculating the diagnostic criteria and methods for carrying out computer-aided tests that take into account the inertial properties of ACPS components in determining their insensitivity can be used for introducing supplements in the Methodical Guidelines for Checking and Testing Automatic Control and Protection Systems of Steam Turbines.

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